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Qualitative Governing Approach of a Spark Ignition Engine using Exhaust Gas Recirculation

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Abstract

A qualitative governing approach of a Spark Ignition (SI) engine incorporating Exhaust Gas Recirculation (EGR) was studied using AVL Boost. SI engines being quantitatively governed by controlling the quantity of charge inducted, leading to high throttling losses. To qualitatively govern the engine, charge purity was reduced by incorporating EGR, thereby reducing the amount of available oxygen in the charge inducted. The effect of charge dilution is to slow down the flame development and propagation, increasing the combustion durations. A simulation model which includes a curve fit equation to predict the combustion duration and start of combustion was created. The engine could be qualitatively governed by a maximum of 19% EGR, to reduce the brake mean effective pressure (bmep) by 20%. This led to 20% reduction in NO_x and CO at full load, with 2.5% increase in brake specific fuel consumption (bsfc), also an increase in HC emissions by 50%.

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Keywords: Qualitative governing, Engine modelling, Exhaust Gas Recirculation, burning velocities, NO_x reduction

1. Introduction

Conventional SI engines are quantitatively governed by controlling the amount of fresh charge being inducted into the cylinders through the throttle valve. This paper aims at governing an SI engine qualitatively by diluting the intake charge, thereby reducing the charge purity. This changes the composition of the intake charge, thereby reducing the amount of Oxygen available for combustion. In this work, Residual Gas content (RG) is used to specify the charge purity. In this paper, the bmep was reduced by introducing EGR, instead of the throttle valve closing, which should lead to a reduction in the pumping losses and hence improvement in thermal efficiency [1,2]. EGR is also seen as a strategy to reduce NO_x emissions, one of the primary air pollutants from an SI engine [1-3]. During combustion in SI engines, since very high peak temperatures are reached, NO_x is primarily formed by the thermal mechanism (Zeldovich Mechanism) [3]. If the flame temperatures during combustion can be reduced, the NO_x formation can also be controlled. The effect of EGR is to increase the heat capacity of burned gases per unit heat added, thereby reducing the maximum flame temperatures [1,3]. There is a limit to the EGR that may be effectively employed, as it slows down the flame development and propagation, thereby increasing the combustion duration, well into the expansion stroke.

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2. Engine modelling and results

A 2.0 litre 4 cylinder, 4 stroke Naturally Aspirated SI engine was used. Stoichiometric air-fuel ratio has been maintained throughout the study, to eliminate the effect of an equivalence ratio on combustion. The engine model, incorporates cooled EGR, tapping for which was taken from the exhaust manifold, downstream of cylinder 3. Vibe 2-Zone combustion model was used. A 2-Zone model enables emission study by considering 2 mass averaged temperature zones; unburned and burned zone [4]. The burned zone was further divided into 10 stratification zones to effectively model NO_x formation. The following 17 species are used: Gasoline (Fuel), O_2, N_2, CO_2, CO, H_2O, H_2, H, O, OH, HO_2, H_2O_2, N, NO, NO_2, NO_3 and N_2O are used to calculate conservation of mass for each species.

The results of the simulation were matched with the available measured engine data by Blair [2] and showing an error of less than 4% for indicated performance. The measured peak firing pressure data indicated 64 and 70 bar at 3600 and 4800 rev/min engine speed respectively [2], whilst simulations gave 64.5 and 71.2 bar, with an error of less than 2%.

3. Combustion calculation

The charge diluent slows the flame development and burning velocities and hence increases the combustion duration, therefore the start of combustion has to be advanced to maintain the Maximum Brake Torque (MBT) timing [1,3]. Abdel-Gayed et al. [5-8] have given correlations to predict turbulent burning velocities based on the laminar burning velocity and flow properties for different flame regime structure. In the current work, wrinkled laminar-flame regime has been used, as evident from experiments for premixed turbulent combustion [3].

3.1. Laminar Burning Velocity

Laminar burning velocity (S_L) is calculated using Metghalchi and Keck correlation [3,9] as follows:

$$S_L = S_{L,ref}\left(\frac{T_u}{T_{u,ref}}\right)^{\gamma}\left(\frac{P}{P_{ref}}\right)^{\beta}(1-2.1RG) \qquad (1)$$

$$S_{L,ref} = B_M + B_2(\emptyset - \emptyset_M)^2 \qquad (2)$$

Where T_u is unburned gas temperature, P is pressure in atm, $T_{u,ref}$ is 298 K, P_{ref} = 1 atm, and,

$$\gamma = 2.18 - 0.8(\emptyset - 1) \qquad (3)$$

$$\beta = -0.16 + 0.22(\emptyset - 1) \qquad (4)$$

Where Φ_M, B_M and B_2 are 1.13, 27.58 cm/s and -78.34 cm/s respectively for Indolene [9], Φ is the equivalence ratio. Values of unburned pressure, temperature, residual gas content and equivalence ratios were obtained from the simulations.

3.2. Turbulent Burning Velocity

Bradley's correlation [10,11] is used to calculate the turbulent burning velocity (S_T),

$$\frac{S_t}{S_L} = 1 + 0.95Le^{-1}\left(\frac{v'_{rms}}{S_L}\frac{l_o}{\delta_L}\right)^{1/2} \qquad (5)$$

Where Le is the Lewis number, v'_{rms} is the turbulence intensity, l_o is the integral length scale, and δ_L is the laminar flame thickness. The Lewis number values were calculated by Abdel-Gayed et al. [5] for a variety of fuels and for gasoline air mixture, Le was reported to be in the range of 1.14 to 4.08, and an average value of 2.61 was assumed. Heywood [1] estimated the integral scale was estimated as 20% of the instantaneous clearance height. The integral scale was calculated at each crankangle and the laminar flame thickness (δ_L) was estimated based on the Kolmorogov scale (l_K), which is reported to be of the order of 0.01 mm at the end of compression [1]. For the wrinkled flamelet regime, $\delta_L \leq l_K$ [12], and a value of δ_L = 0.01 mm is used in the current work. Turbulence intensities were calculated following Prucka et.al. [13], taking into account the engine speed, turbulence decay during compression, bmep, and combustion phasing.

Fig.1 shows the measured combustion duration [2] and the turbulent burning velocities calculated using Eq. 1 and 5, and a curvefit equation was obtained for this engine,

$$t_{CDUR} = 0.185.S_t^{-1.6} \qquad (6)$$

The constants in Eq. 6, will depend on the engine configurations. Using combustion duration prediction from Eq. 6, the start of combustion (SOC) and the shape of heat release curves are then calculated based on the Vibe function. The error of 10% between the predicted and actual combustion duration is deemed reasonable for the limited measured data available.

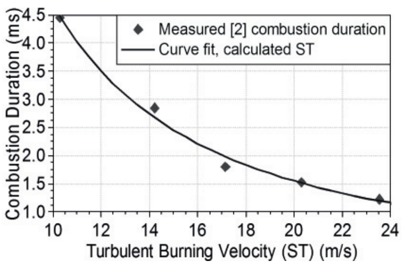

Fig. 1: Combustion Duration against S_t

4. Qualitative Governing using EGR

Fig. 2 shows that S_t/S_L increases with an increase in EGR. The S_L decreases with an increase in residual gas content, hence both v'_{rms}/S_L and S_t/S_L increase. It must be noted that the turbulent burning velocities also reduce slightly.

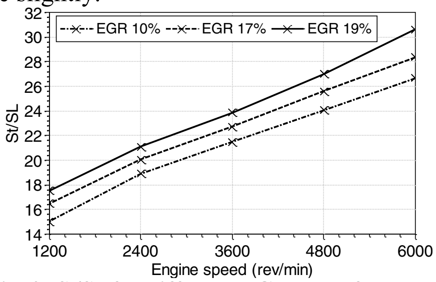

Fig. 2: S_t/S_L for different EGR mass flow rates

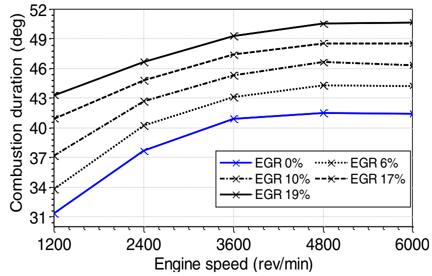

Fig. 3: Combustion duration with EGR

Fig. 3 shows the combustion durations increasing with an increase in EGR, calculated using Eq. 6, with residual gas content, there is a limit on the amount of EGR that can be incorporated in the cylinders, without causing detrimental effects on combustion performance, which in this case was found to be about 20%. Fig. 4 shows that the residual gas content increases by 3 times for maximum EGR of 19%.

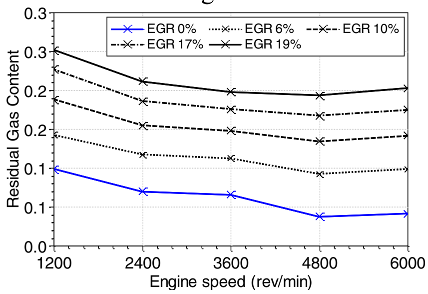

Fig. 4: Residual gas content with EGR

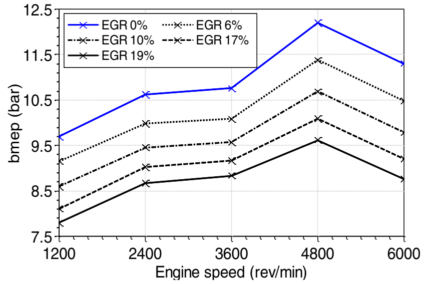

Fig. 5: bmep for different EGR mass flow rates

Fig. 5 evidently shows that up to 20% reduction in bmep can be achieved at full load using 19% EGR.

Fig. 6 shows that maximum bsfc increased only about 2.5% with a reduction in 20% bmep using 19% EGR.

From Figs. 5 and 6, it can be concluded that for the same load, the bsfc is reduced with increase in EGR. Also in Fig. 6, the 10 bar bmep curve is shown as an example, indicated that using EGR to qualitatively govern the engine is more efficient.

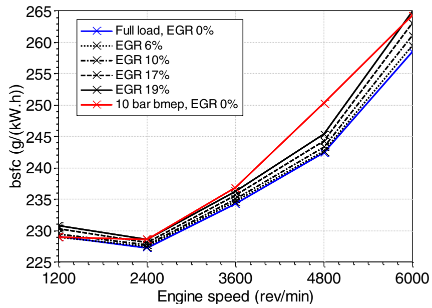

Fig. 6: bsfc comparison with bmep and EGR

5. Species Emission

Fig. 7 shows a reduction in the peak burned gas temperatures, by 200-300 K (about 10%) for 19% EGR for different engine speeds.

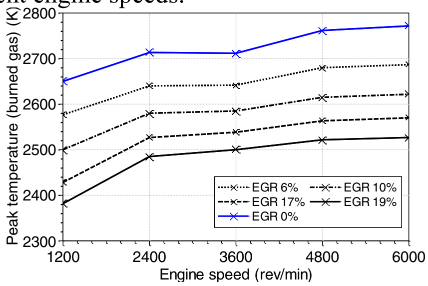

Fig. 7: Peak burned gas temperatures with EGR

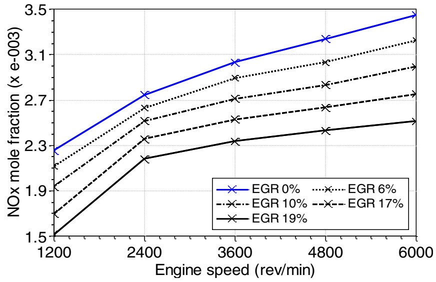

Fig. 8: NO_x species mole fraction with EGR

Fig. 8 shows NO_x reduced due to the reduction in the peak temperatures generated during combustion, with NO_x by a between of 20% to 33%. The simulation also showed brake specific NO_x (g/kWh) were reduced by 70% for 19%. Fig. 9 shows a reduction in CO which can be explained due to the reduced dissociation of CO_2 as the peak cylinder temperatures reduce [3].

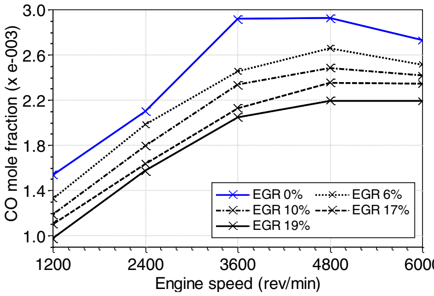

Fig. 9: CO mole fraction with EGR

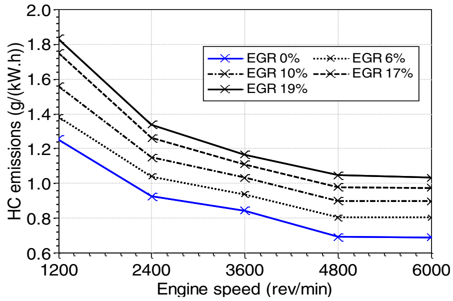

Fig. 10: HC emissions with EGR

Fig. 10 shows increase in HC with increase in EGR. This can be explained by the reduction in the Oxygen content in the charge.

6. Conclusions

1. A simulation model which includes a curve fit equation to predict the combustion duration and start of combustion was created.
2. With an increase in charge dilution, both S_L and S_t are reduced, whilst the reduction is S_L is predominant leading to an increase in S_t/S_L.
3. For a maximum EGR of 19%, the bmep reduced by 20%, with 2.5% increase in bsfc, and reduction in NO_x and CO between 20% and 33%.
4. Increase in EGR, causes a reduction in peak unburned temperatures causing a reduction in NO_x and CO. However the HC emissions have increased by about 50%.
5. Reducing HC emissions can be investigated further by studying the effect of leaner mixtures on combustion.

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